# A LABORATORY INVESTIGATION OF THE ADAPTABILITY OF THE HEAT PUMP TO BATCH DRYING OF SHELLED CORN

by

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B. S., Kansas State College of Agriculture
and Applied Science, 1952

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A THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Agricultural Engineering

KANSAS STATE COLLEGE
OF AGRICULTURE AND APPLIED SCIENCE

1953

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# INTRODUCTION

The first production of artificial refrigeration by use of a machine took place over one hundred years ago, the earliest recorded patent for such a machine being issued in Great Britain in 1834. A few years later, in 1852, Lord Kelvin pointed out that the heat rejected from a refrigeration system could be employed to warm a space. The term "heat pump" is now generally applied to refrigeration equipment used for heating purposes.

In recent years considerable emphasis has been placed on the use of the heat pump for the heating and cooling of homes and office buildings, and commercial units for this purpose are now readily available. Industry also offers many opportunities for the economic utilization of the heat pump. A number of manufacturing processes require the control of temperature and humidity within close limits for the production of quality products. The list of agricultural processes fitted to the heat pump includes evaporation, distillation, concentration, and drying.

An increased interest in the agricultural potentialities of the heat pump has prompted several writers to suggest the possibility of using the heat pump for the conditioning of farm crops. Modern harvesting methods and the desire for early harvesting have resulted in a need for an economical method of drying farm grains.

High moisture content grain is usually prepared for storage by forcing air through the grain. The grain moisture content, the grain and air temperature, and the humidity of the air are factors that determine the amount of moisture a given quantity of air can remove from a product. The inherent features of the heat pump make it readily adaptable for controlling two of these factors, air temperature and humidity. Consequently, there may be distinct advantages in using the heat pump for the drying of grain as compared to the heated air driers now on the market.

When heated air is forced through the grain in batch driers, the vapor-pressure difference between the drying air and the grain is greatest at the point where the air first enters the grain. The grain gives up moisture to the air, and the drying capacity of the air is lowered. The vapor-pressure difference becomes less and less as the air moves through the grain, and near the exhaust side of the bin the rate of drying will be slower. This results in a moisture gradient in the grain which remains after the drying is completed, and it often becomes desirable to mix the batch before storage to obtain a more uniform moisture content. The heat pump appears to offer a method of conditioning all of the product to the same moisture content since theoretically the temperature and humidity of the drying air can be maintained in equilibrium with the desired final moisture content.

## REVIEW OF LITERATURE

Barre (2) stated that the transfer of moisture in the form of vapor is distinctly a flow problem and that the force tending to bring about flow is the vapor-pressure difference existing between the moisture-holding material and its surroundings. He further stated that the rate at which a material will lose its moisture depends primarily on the magnitude of the vapor-pressure difference and the permeability of the material to the flow of water vapor.

Barre, p. 249, concluded:

> It is apparent from some of the above discussion that, in order to obtain large vapor-pressure differences, the material to be dried must be heated to increase its vapor pressure and that the pressure of the surrounding water vapor be kept low, if rapid drying is to be accomplished.
>
> In applying this concept to artificial drying with forced heated air, as in grain drying, the essential purpose which the heated air accomplishes in passing through the grain is the heating of the grain to increase its vapor pressure.

Davis (3), pp. 8-9, made the following comments:

> . . . the air plays the additional function in drying of supplying the latent heat of vaporization plus an additional amount equivalent to the heat of wetting. The moisture in the grain is diffused through the grain kernel to the moving air stream and carried away in the form of water vapor. The grain itself may undergo a sensible heat change and the equivalent of the "heat of wetting" is absorbed. The total heat of the air will thus be modified by the algebraic sum of the sensible heat change in the grain and the equivalent of the heat of wetting.

Hukill (6), p. 340, in summarizing the results of a study of the basic principles in drying corn and grain sorghum, wrote:

> The maximum heat available for drying is the sensible heat of the air between its initial dry-bulb temperature and the dry-bulb temperature corresponding to the wet-bulb of the entering air at the equilibrium relative humidity of the wet grain.
>
> When artificial heat is used, drying is generally more economical at high temperatures than at low ones. This effect appears in two ways. First, as heat is added to saturated air, the ratio of the increase in wet-bulb depression to the rise in dry-bulb temperature becomes larger as more heat is added. That is, the ratio of available heat to heat supplied becomes larger. If the atmospheric air is less than saturated, there is some free available heat in the atmosphere which may change this relation. Second, as the temperature rises the time required for a drop in moisture to the halfway point is lessened and the drying period may be shortened without losing efficiency.

Trent (11), p. 116, gave a somewhat simpler explanation:

> The amount of moisture which a given quantity of air can remove from the grain bed is determined by two factors, the temperature and the initial moisture content of the entering air. Therefore, it is desirable to use air with as high a temperature and with as low a moisture content as possible.

The temperature of the drying air is limited by the biological characteristics of the grain. Maximum drying temperatures have been determined for various grains for optimum germination, acceptable animal feed, and proper milling qualities. These temperatures have been summarized and reported in table form by Baker (1). As reported by the B.P.I.S.A.E., U.S.D.A., the maximum drying air temperature for acceptable milling qualities of corn is 130° F and for proper seed germination 110° F.

Much of the grain drying done on the farm and by commercial establishments is accomplished by blowing heated air through the grain until its moisture content is reduced to the point

where safe storage is assured. Most of the heated air driers now available are of the combustion type using oil or a similar fuel for the heat source. These driers operate satisfactorily and are economically feasible; however, certain thermal inefficiencies are involved with this type of drying system. The principal loss is the exhausting of latent and sensible heat to the ambient air via the drying air which is forced through the grain. Overdrying of the grain close to the air entrance takes place, and the efficiency of the oil fired drier decreases as the ambient air temperature decreases.

With the above inefficiencies in mind, Davis (3), p. 3, stated:

> It would seem desirable then to utilize equipment which can operate virtually independently of ambient conditions of both temperature and relative humidity. At the same time it is desirable to reclaim the unutilized sensible heat as well as the latent heat of evaporation which is normally discharged to the atmosphere and is therefore unavailable for further drying. This may be accomplished by the heat pump operating in a closed cycle.

Trent (11), p. 116, explained the operation of a heat pump grain-drying system using recirculated air (Fig. 1) as follows:

> Warm air leaving the condenser of the heat pump passes through the grain bed under influence of the fan and picks up moisture from the grain. The air then passes on to the evaporator coil where its temperature is reduced below the dew point and a considerable amount of moisture is removed. Leaving the evaporator the air flows by the compressor and electric motor where it picks up a small amount of heat. It then passes over the condenser. The condenser raises the air to a moderately high temperature in which condition it again enters the grain bed and begins another cycle through the system.

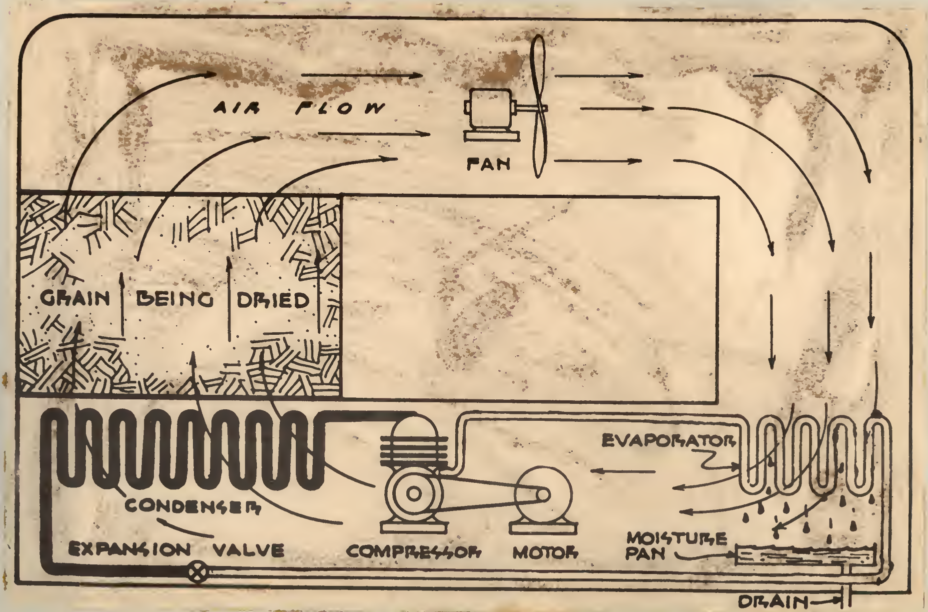


Fig. 1. Schematic diagram of a heat pump grain drying system.

The agricultural applications of the heat pump suggested by Zastrow (12) include the curing and storing of perishable food crops. Zastrow, p. 204, also made the following statements concerning the heat pump:

> The development of new applications will depend at least partly on how accurately the capabilities of this equipment are evaluated. Limitations are its relatively high first cost, the need for an adequate heat source and its inability to furnish heat economically at high temperatures. On the other hand, the heat pump can offer advantages in cost over other methods of heating where the heat is used at moderate temperatures, where an adequate source of heat is available, and where the utilization factor is such that a higher first cost is justified. Furthermore, it is well suited for easy control or automatic operation, it does not produce dust or smoke, and it is able to provide cooling as well as heating.

Downs (4) in a discussion of new approaches to crop drying problems stated:

> . . . the closed-cycle heat pump appears to have the following advantages: (1) it eliminates the fire hazard, (2) little or no attendance is required, (3) the equipment can be adjusted to prevent over-drying, regardless of weather conditions, and (4) drying can be carried out effectively at relatively low temperatures.

Although limitations may become apparent later, it appears that there may be some distinct advantages in favor of drying grain with a heat pump system. From his comparative analysis of the heat pump and oil fired drier Davis (3), pp. 45-46, drew the following conclusions:

> 1. When used in air-conditioning for the drying of grain in a closed air cycle, the heat pump operates most efficiently if: (1) The compressor discharge and suction operating temperatures are as near each other as possible (for heating and dehumidifying the drying

air). (2) The two temperatures are the maximum compatible with compressor design and consistent with an air temperature which does not damage the viability of the drying grain.

2. Operation may be carried on virtually independently of ambient air conditions.

3. Heat is more largely utilizable for drying with high temperature than from low temperature air.

4. For cold weather drying, the heat pump dries grain for approximately 20 per cent less operational energy cost than the conventional oil fired drier. The energy cost per bushel for drying with the heat pump is approximately 1.4 cents per bushel.

5. As depths of grain increase, the pressure that the blower must furnish becomes larger. This is particularly true for the heat pump closed cycle drier as efficient operation of the compressor involves the use of large quantities of air. With the heat pump, operation on a competitive total energy basis is possible for depths of grain up to 2 1/2 feet under present cost conditions, but this would vary with the existing rates for oil and electricity.

6. In view of the economic factors involved in the cost of equipment and servicing and the manifold problems of adapting and utilizing this equipment for grain drying on the farm, further study regarding these factors must be made before the construction and installation of such equipment is recommended.

Davis' study was made on the basis of continuous type drying. However, he also pointed out that the heat pump could operate more favorably than the oil fired drier in a batch type drying system since the heat is continuously reclaimed for further drying. He recommended that this phase would be worthy of exploration, and hence the laboratory investigation concerned in this thesis was made.

## THE INVESTIGATION

This investigation was initiated to study the possibilities of drying grain with the heat pump. Although thermodynamically there was little doubt that successful drying could be accomplished with a heat pump system, more information was needed before a field installation could be properly designed. In view of this fact, laboratory testing was chosen in preference to field testing. It also seemed desirable that this preliminary work be conducted using a batch drier since most of today's drying installations are of this type.

An objective of this investigation was to attempt to obtain a uniform final moisture content throughout the batch by supplying air at the equilibrium conditions corresponding to the final moisture percentage desired. Another objective was to study the effects of different relative humidities of the drying air upon the drying rate, time of drying, and the energy requirements for drying.

Because of the dry harvest season of 1952, difficulties were experienced in locating high moisture content grain. The corn used in the tests had an initial moisture content of approximately 15 to 16 percent (WB). All the tests were conducted on naturally wet grain with the exception of Test No. 3. The grain used in this test was rewetted grain that had been dried in the previous test.

## LAYOUT OF TEST EQUIPMENT

The experimental recirculated air heat pump grain drying installation was assembled in the laboratory as shown in Fig. 2 and in Plate I. Plywood construction was used for the test bin and the heat pump enclosure. After a preliminary test, the heat pump enclosure and the metal air return duct were insulated with blanket-type balsa insulation to reduce the heat loss from the system. For the tests the upper two sections of the test bin were removed and approximately two feet of grain was placed in the bottom section.

## INSTRUMENTATION

Thermocouples were installed for measuring the various temperatures. Temperature readings were taken on a portable indicating potentiometer and periodically checked with a multiple-point recording potentiometer.

### The Air System

The wet bulb temperature[1] and the dry bulb temperature of the recirculated air were recorded for the following points: (a) before the air entered the grain, (b) after the air passed through the grain, and (c) after the air passed over the evaporator. The temperatures were plotted on a psychrometric chart to determine the relative humidity at each point.

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[1] For details of the wet bulb temperature thermocouple construction see: "Instrument News," Karl Norris, Editor, Agricultural Engineering, October 1952, 33(10):644.

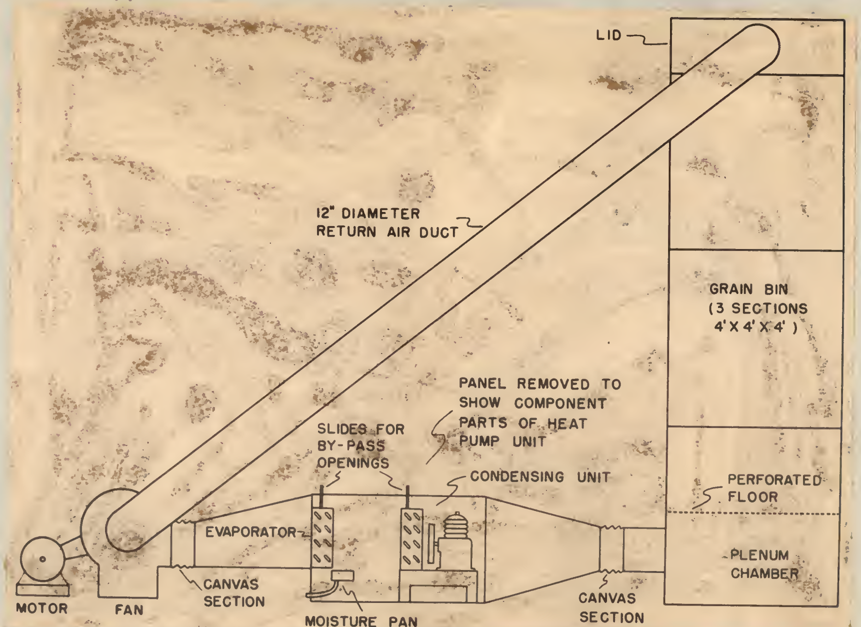



Fig. 2. Scale drawing of an experimental recirculated air heat pump grain drying installation.

## EXPLANATION OF PLATE I

View of the experimental recirculated air heat pump grain drying installation.

Components of installation:

1. Test bin.
2. Insulated duct (for air return).
3. Enclosure for condensing unit and evaporator.
4. Fan.
5. Fan motor.

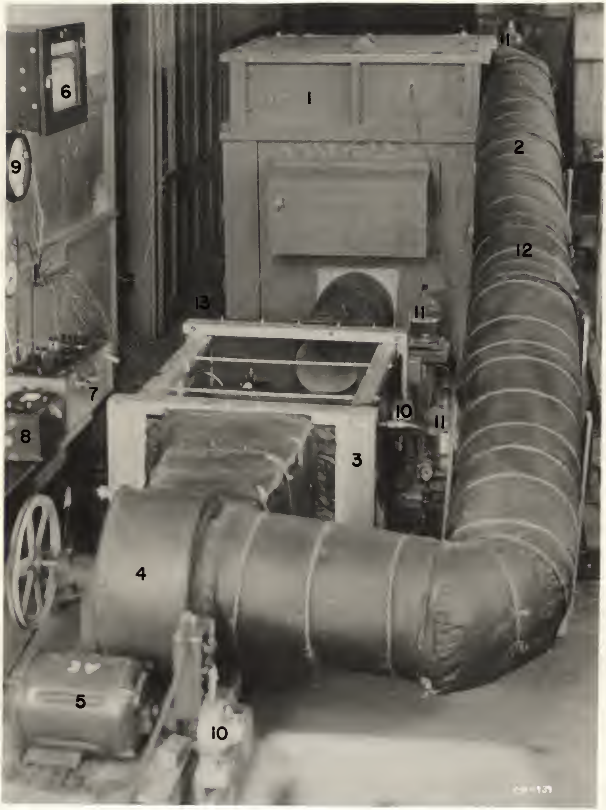
Instrumentation:

6. Multiple-point recording potentiometer.
7. Portable indicating potentiometer.
8. Meter box for Tag-Heppenstall moisture meter.
9. Recording compound pressure gauge.
10. Kilowatt hour meter.
11. Water jug for wet bulb thermocouple.
12. Location of air flow nozzle in duct.
13. Platform scale.

Not included in picture:

a. Refrigerant flow meter.
b. Slanting-leg manometer (for static pressure).
c. Hook gauge (for pressure differential across air flow nozzle).

PLATE I



A hook gauge was used to measure the pressure differential across a calibrated flow nozzle placed in the return air duct. The quantity of air circulated was then calculated. A slanting-leg manometer was used to measure the static pressure. A positive static pressure reading was obtained in the plenum chamber under the grain, and a negative reading was obtained in the lid section above the grain. The total static pressure was the numerical sum of the two readings.

The electric motor and fan available were not entirely suitable for the air movement desired for the tests. Consequently, the air flow energy requirements as measured with the kilowatt hour meter on the fan motor were not applicable to the analysis. As a result, the energy requirements for maintaining the air circulation were calculated by using the measured values of air flow and static pressure.

## The Refrigerant System

The refrigerant (Freon 12) temperatures were obtained by soldering thermocouples to the copper refrigerant lines. After the soldering was completed, the thermocouples were insulated. These thermocouples were placed at the following points: (a) near the compressor on the discharge line, (b) before the expansion valve, (c) after the expansion valve, and (d) near the evaporator on the suction line.

A recording compound pressure gauge kept a continuous record of the condensing pressure and the suction pressure.

Indicating pressure gauges were installed for reading the head pressure and the evaporator pressure. A manual shutoff valve was installed in the suction line and operated as a manual evaporator pressure regulator for maintaining the desired evaporator pressure.

The refrigerant flow was measured with a rotameter area type flow meter, and a kilowatt hour meter recorded the energy requirements of the three-quarter horsepower condensing unit.

## The Grain System

Thermocouples were placed in the grain for measuring the grain temperature. These thermocouples were located as nearly as possible to coincide with the depths from which the grain samples were obtained. Although the air movement through the grain undoubtedly prevented the thermocouples from reading the true grain temperature, an indication of the heating of the various layers of grain was obtained.

Platform scales placed under the test bin were used to determine the weight of the grain placed in the bin. The decrease in weight read on these scales during the tests was considered to be the weight of the moisture removed from the grain.

Access holes in the test bin lid permitted the use of a grain probe for obtaining grain samples for moisture content determinations. The samples were placed in metal containers (previously weighed), weighed, placed in an oven at 130° F for

24 hours, and then reweighed. The wet basis moisture percents were then determined. For later calculations these percents were converted to a dry basis. During the tests a portion of each sample was put through a Tag-Heppenstall moisture meter for immediate checking of the moisture content.

The relative humidity of the air leaving the grain and the approximate temperature of the grain were used with equilibrium moisture content data (10) to check the moisture content of the top layer of grain. This provided a reasonably accurate check as the grain and air approached equilibrium.

## TEST PROCEDURE

Prior to the starting of a test, the procedure was as follows:

(a) Approximately 25 bushels of shelled corn was placed in the test bin.

(b) The grain temperature thermocouples were placed at their proper depths.

(c) The test bin lid was sealed.

(d) A sample was taken for determining the initial moisture content of the corn.

(e) The wet bulb temperature thermocouples were checked for proper operation.

(f) Kilowatt hour meter readings were taken.

(g) The time was recorded.

As soon as the unit was started, the evaporator pressure valve was adjusted to give the desired evaporator pressure and the expansion valve was adjusted to give the desired refrigerant flow for the particular test. This was a simultaneous adjustment as the changing of one affected the other. Minor adjustments were then made during the test to maintain a constant evaporator pressure and refrigerant flow. The pressure differential for the air flow calculation was read on the hook gauge, and the static pressure was obtained with the slanting-leg manometer. The speed of the fan had been previously set to maintain an air circulation of approximately six cubic feet of air per minute per bushel of grain.

During the starting period two men were present, and each checked all readings and adjustments. After the test was under way and the initial readings were obtained, it was necessary for only one man to remain. A constant watch was maintained over the unit for approximately the first 40 hours of operation.

Grain samples for moisture determinations were taken every three hours. The following data were recorded every two hours:

(a) Temperatures (air, refrigerant, and grain).

(b) Weight of grain moisture removed.

(c) Kilowatt hour readings.

(d) Head pressure.

(e) Suction pressure (recorded on continuous chart).

(f) Evaporator pressure (held constant).

(g) Condensing pressure (recorded on continuous chart).

(h) Refrigerant flow (held constant).

(i) Hours of operation of test.

The rate of drying had decreased to a slow rate after approximately 40 hours of operation, and adjustments of the unit were no longer necessary. The intervals between the data recordings were lengthened, and the constant vigil was discontinued. Only slight variations occurred in this procedure from test to test.

## TEST RESULTS

Five grain conditioning tests were conducted. The initial test will be designated as the "preliminary test." The preliminary test will not be discussed as it served only as a guide for the following tests. However, it was apparent from the preliminary test that different relative humidities of the drying air could be obtained by varying the evaporator pressure (Fig. 3); and the decision to insulate the return air duct and the plywood housing was made after reviewing the results of this test. There was evidence that the heat loss from the system could be decreased considerably and a higher drying air temperature obtained by insulating these sections. The tests that followed proved that this was a valid conclusion.

The four tests discussed will be designated as Test Nos. 1, 2, 3, and 4. A summary of the tests is given in Table 1.

### The Air System

Curves of the dry-bulb temperature and relative humidity of

the air entering and leaving the grain are shown in Figs. 4, 5, 6, and 7. In each test the relative humidity of the air entering the grain was lower than the relative humidity corresponding to the evaporator pressure as indicated by the curve of Fig. 3. Figure 3 is based on an air temperature of $90^{\circ}$ F; and as higher air temperatures were obtained in the tests, lower relative humidities resulted.

Difficulty was experienced in obtaining the desired operating conditions for Test No. 4. The unit was operated for six hours before the difficulties were corrected; and although five days elapsed before the test was started again, the initial temperature of the grain was $10^{\circ}$ F higher than in the previous tests. This may account in part for the high air temperature obtained in Test No. 4.

The heat from the fan motor was dissipated to the atmosphere. However, the energy requirements for maintaining the air flow were so small (Appendix) that the resulting heat gain would have been negligible had the fan motor heat been utilized in the system.

## The Refrigerant System

During Test Nos. 1 and 2 the refrigerant flow was maintained at a constant value, and the evaporator was operated at a different pressure for each test. These evaporator pressures were repeated with a higher refrigerant flow in Test Nos. 3 and 4. Although the evaporator pressure valve was not restricting the suction line during Test Nos. 1 and 3, it will be noted that the

evaporator pressure was higher in Test No. 3 (Table 1) as a result of the different operating conditions.

The theoretical coefficients of performance in terms of the enthalpies were calculated for comparative purposes and are reported in Table 1 as the average of the coefficients calculated at three points during the operation of each test (Appendix). The enthalpies were read from a pressure-enthalpy diagram by using the measured values of the pressures and temperatures of the refrigerant.

## The Grain System

Each test was terminated when the thermocouples placed in the grain indicated that the entire grain mass was at approximately the same temperature. Figures 8, 9, 10, and 11 show that the rate of moisture removal decreased to a low value after 50 hours of operation; consequently, little moisture was removed during the period required to bring the upper layer of grain up to a temperature equal to that of the lower layers.

The drying rate curves (Figs. 12, 13, 14, and 15) indicate the rate of drying of the various layers of grain. The ordinate is a grain moisture ratio and represents any combination of initial and final moisture content, 0 being the equilibrium moisture content corresponding to the entering air condition and 100 being the initial moisture content. The grain moisture ratio is equal to:

$$\frac{M - M_E}{M_0 - M_E} \times 100$$

where:

$M$ = Moisture content percent at any time.

$M_E$ = Equilibrium moisture content percent corresponding to the entering drying air condition.

$M_0$ = Initial moisture content percent.

The energy requirements for the tests (Table 2) are given for the same moisture limits for each test except Test No. 3. The limits for Test No. 3 are slightly lower because of the lower initial moisture content of the grain used in this test.

Table 1. Summary of grain conditioning tests.

| | Test No. 1 | Test No. 2 | Test No. 3 | Test No. 4 |
|---|---|---|---|---|
| Date started | 12/7/52 | 12/29/52 | 1/18/53 | 2/5/53 |
| Date ended | 12/11/52 | 1/2/53 | 1/21/53 | 2/8/53 |
| Hours of operation | 83 | 93 | 74 | 73 |
| Refrigerant flow (lbs./hr.) | 110 | 110 | 147 | 147 |
| Evaporator pressure (lbs./sq. in. gauge) | 27.5 | 50 | 35 | 50 |
| Air flow (cfm/bu. based on dry weight of 47.32 lb./bu.) | 6.16 | 6.24 | 5.91 | 6.14 |
| Static pressure (inches of water) | .13 | .12 | .12 | .12 |
| Average initial moisture percent, dry basis (oven dried sample) | 18.04 | 20.00 | 16.93 | 17.76 |
| Average final moisture percent, dry basis (oven dried sample) | 8.65 | 9.87 | 8.70 | 9.19 |
| Dry matter weight of grain (lbs.) | 1130.2 | 1194.5 | 1188.7 | 1223.7 |
| Total weight of moisture removed, scale weight (lbs.) | 105 | 109.5 | 101.4 | 98.9 |
| Total condensing unit energy, metered (kw-hrs.) | 86 | 97 | 89.5 | 94.5 |
| Total fan energy, calculated (kw-hrs.) | .4 | .4 | .3 | .3 |
| Average coefficient of performance (heating) | 4.68 | 4.56 | 4.09 | 4.62 |
| Average coefficient of performance (cooling) | 3.68 | 3.56 | 3.09 | 3.62 |
| Maximum drying air temperature obtained (°F) | 109.6 | 101.5 | 112.6 | 117 |
| Average initial temperature of grain (°F) | 54 | 49 | 47 | 61 |

Table 2. Energy requirements for grain conditioning tests.

| | Test No. 1 | | | Test No. 2 | | | Test No. 3 | | | Test No. 4 | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Range of decrease of average moisture percent (dry basis) | 17.5 to 15.5 | 17.5 to 13.5 | 17.5 to 11.5 | 17.5 to 15.5 | 17.5 to 13.5 | 17.5 to 11.5 | 16.9 to 14.9 | 16.9 to 12.9 | 16.9 to 10.9 | 17.5 to 15.5 | 17.5 to 13.5 | 17.5 to 11.5 |
| Moisture removed (lbs.) | 22.6 | 45.2 | 67.8 | 23.9 | 47.8 | 71.7 | 23.8 | 47.6 | 71.3 | 24.5 | 48.9 | 73.4 |
| Operation time required (hrs.) | 6.0 | 13.4 | 27.3 | 11.4 | 24.2 | 50.2 | 9.4 | 18.0 | 31.6 | 11.4 | 19.4 | 29.0 |
| Condensing unit energy (kw-hrs.) | 7.6 | 14.8 | 29.1 | 11.8 | 26.5 | 52.0 | 12.5 | 23.5 | 39.1 | 13.3 | 23.5 | 36.7 |
| Fan energy (kw-hrs.) | .1 | .1 | .1 | .1 | .1 | .2 | .1 | .1 | .1 | .1 | .1 | .1 |
| Total energy (kw-hrs.) | 7.7 | 14.9 | 29.2 | 11.9 | 26.6 | 52.2 | 12.6 | 23.6 | 39.2 | 13.4 | 23.6 | 36.8 |
| Kw-hrs./lb. moisture removed | .341 | .330 | .432 | .498 | .556 | .728 | .530 | .496 | .550 | .547 | .482 | .502 |
| Kw-hrs./bu. of grain dried based on dry weight of 47.32 lb./bu. | .322 | .624 | 1.22 | .472 | 1.05 | 2.07 | .502 | .940 | 1.56 | .518 | .913 | 1.42 |

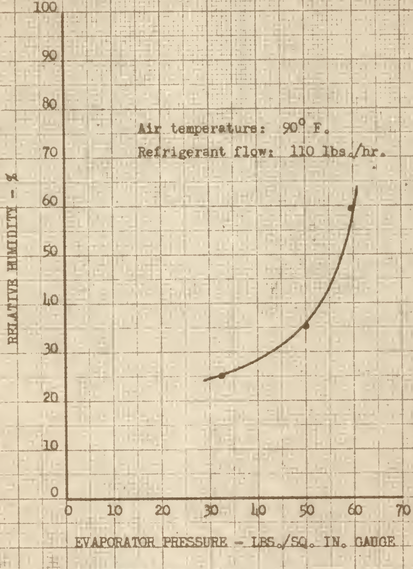



Fig. 3. Relative humidity obtained at varying evaporator pressures.

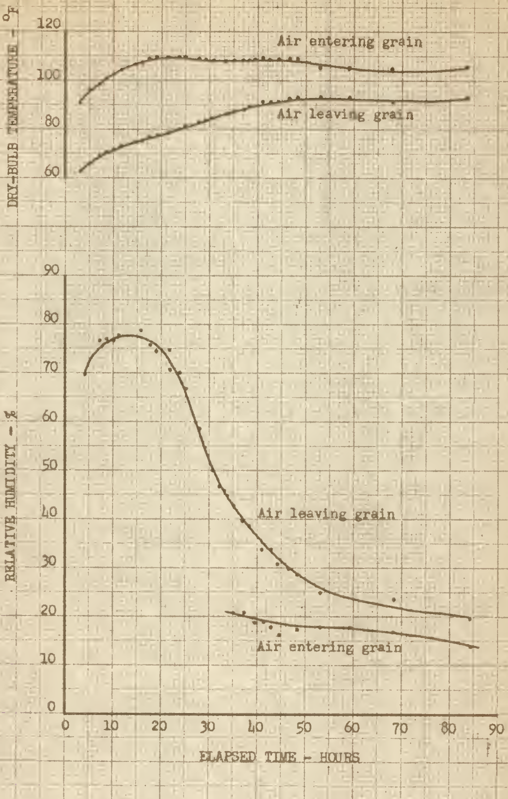



Fig. 4. Temperature and relative humidity of air during Test No. 1.

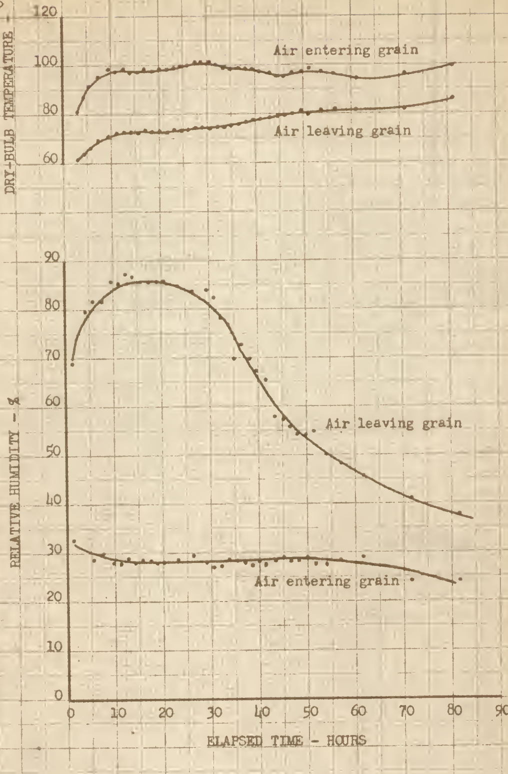



Fig. 5. Temperature and relative humidity of air during Test No. 2.

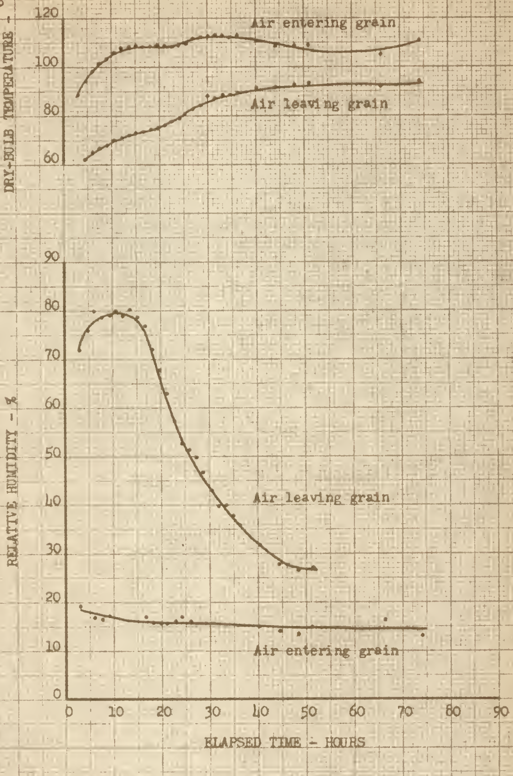



Fig. 6. Temperature and relative humidity of air during Test No. 3.

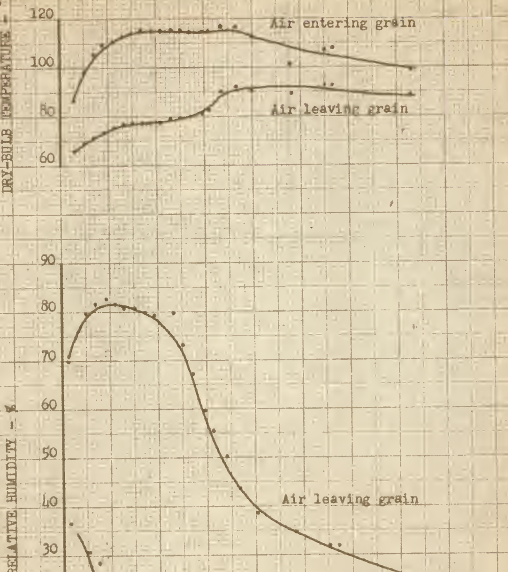



Fig. 7. Temperature and relative humidity of air during Test No. 4.

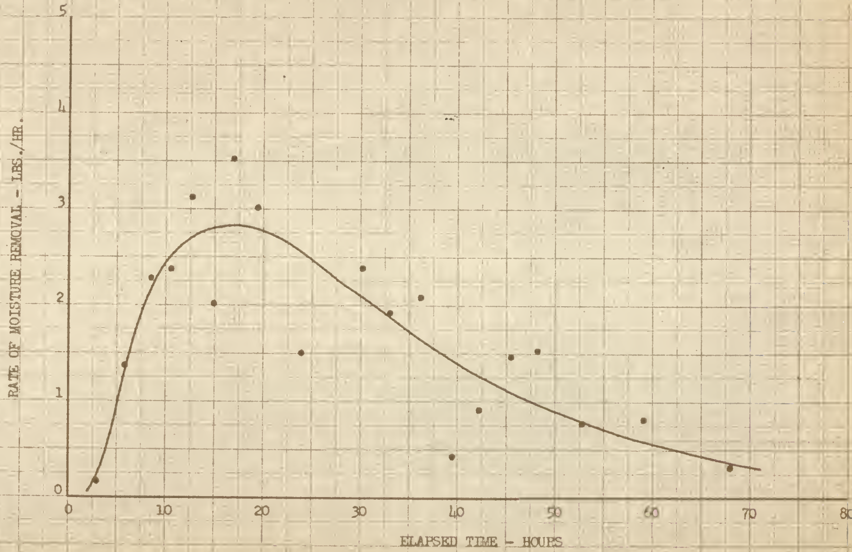



Fig. 8. Rate of moisture removal for Test No. 1.

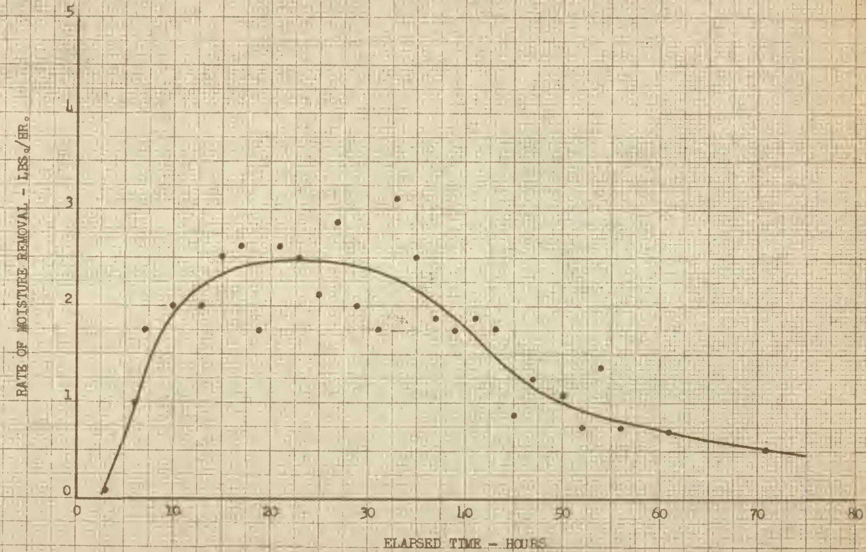



Fig. 9. Rate of moisture removal for Test No. 2.

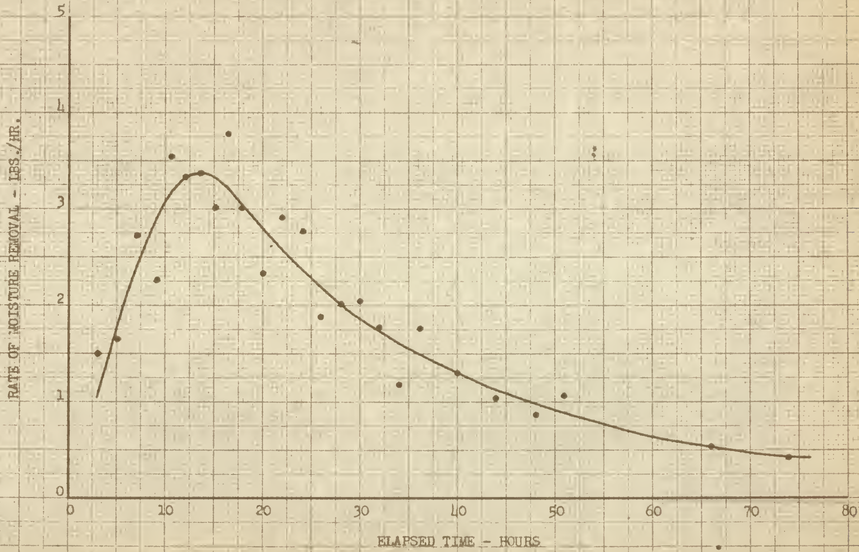



Fig. 10. Rate of moisture removal for Test No. 3.

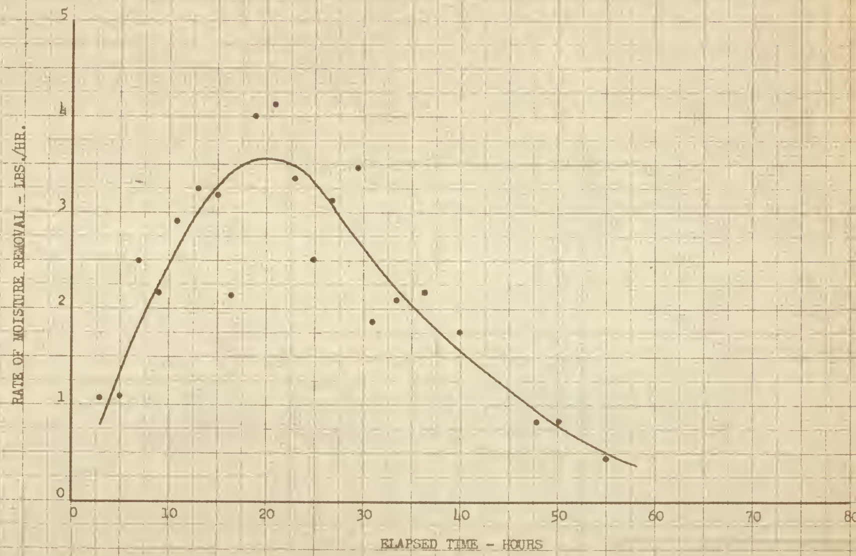



Fig. 11. Rate of moisture removal for Test No. 4.

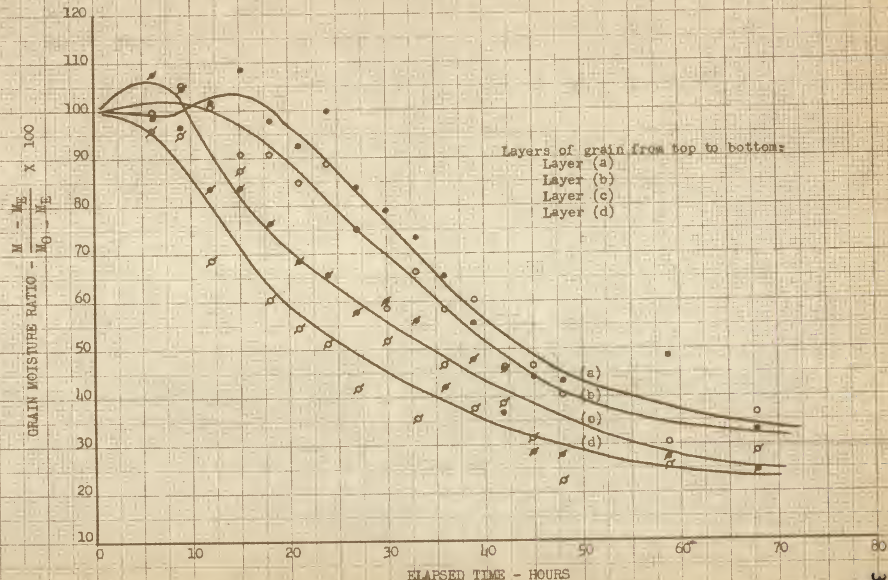



Fig. 12. Drying rate curves for Test No. 1.

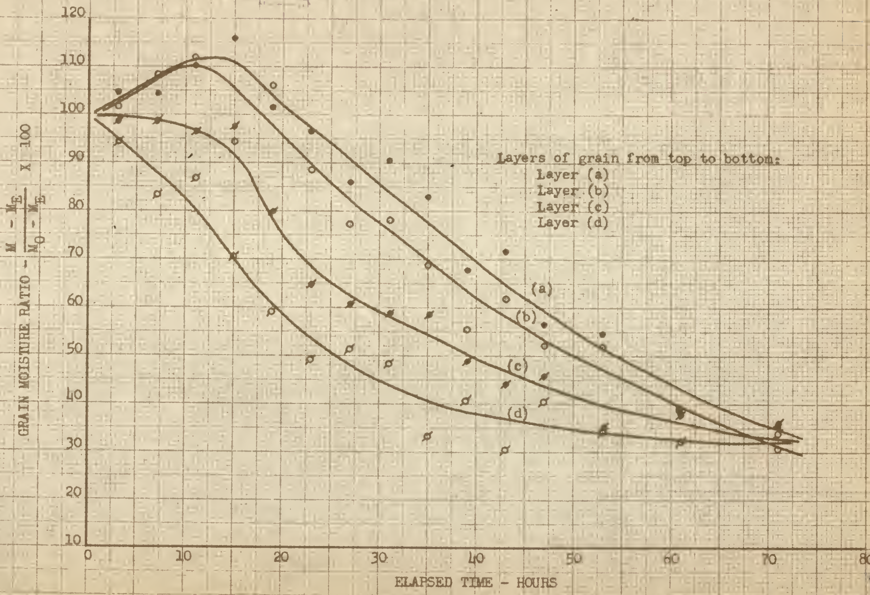



Fig. 13. Drying rate curves for Test No. 2.

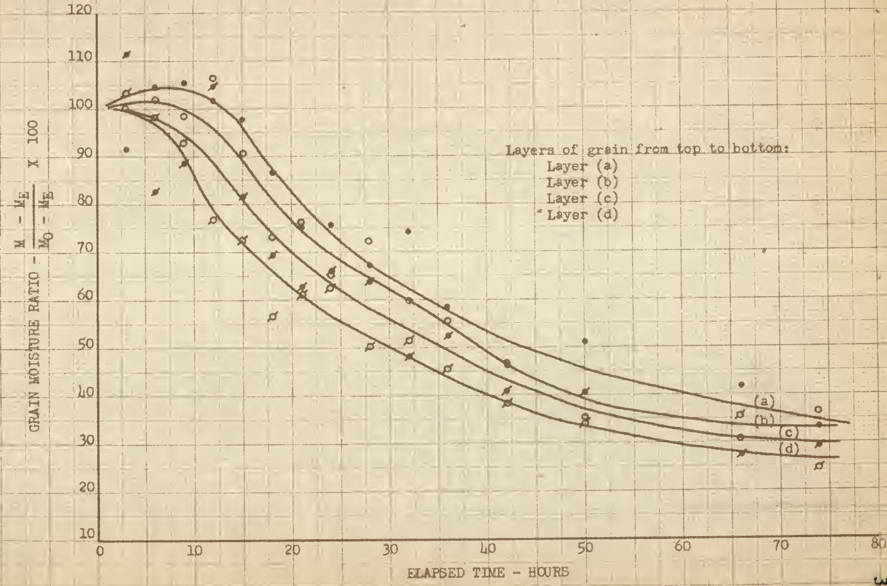



Fig. 14. Drying rate curves for Test No. 3.

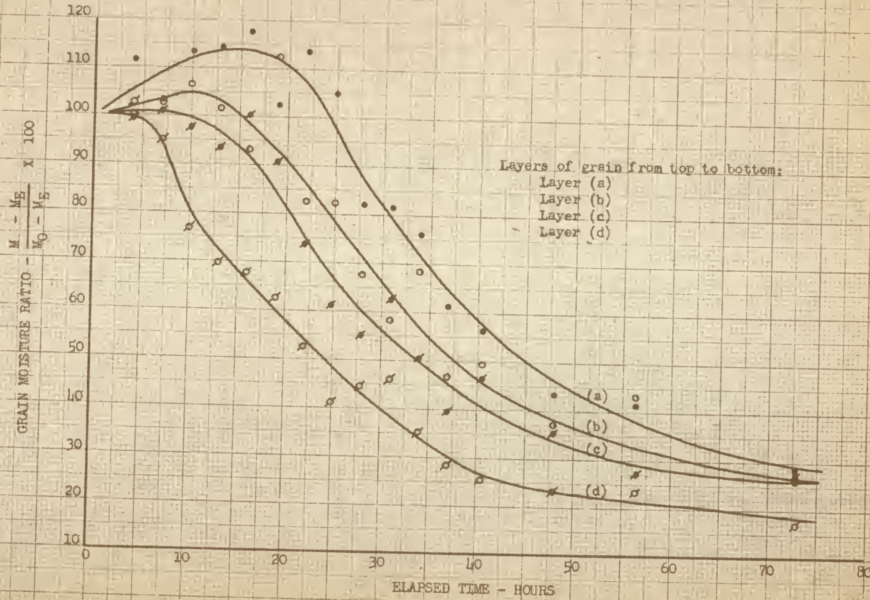



Fig. 15. Drying rate curves for Test No. 4.

# DISCUSSION OF TEST RESULTS

## The Air System

The drying rate curves indicate that during the early part of the tests some of the moisture removed from the bottom layers was redeposited in the top layers. This occurred when the warm, moist air from the lower layers came in contact with the upper layers of cool grain. During this period the air leaving the grain approached saturation; however, after drying started in the top layer, the relative humidity decreased and the full moisture carrying capacity of the air was not used. The air leaving the grain was in equilibrium with the grain during the period that drying was taking place in the top layer; therefore, the air was picking up the maximum amount of moisture possible. However, there was a decrease in the ratio of the sensible heat used in the drying process to the sensible heat supplied to the drying air. If this ratio is to approach unity for a drying system in which the air is dehumidified by cooling before it is heated, the air leaving the grain must approach saturation. To maintain a high relative humidity of the air leaving the grain during the entire drying period, the layer of grain with which the air last comes in contact must continuously have a high moisture content. This did not occur in the batch type drier, and thus the preceding discussion points to the advantage of a higher thermal efficiency which can be maintained in a continuous type drier. Thermal efficiency as used here is defined as the ratio of the

sensible heat used for drying as the air passes through the grain to the sensible heat supplied to the drying air.

The effect of the relative humidity of the drying air on the rate of drying is shown in Test Nos. 1 and 2. In Test No. 2 the relative humidity of the air entering the grain was 10 percent higher than in Test No. 1. The comparisons made in Table 2 show that the operation time required to reduce the moisture content from 17.5 to 11.5 percent (DB) was nearly doubled for Test No. 2 as compared to the operation time required for the same moisture limits in Test No. 1. The energy requirements for these moisture limits were 2.07 kw-hrs./bushel for Test No. 2 and only 1.22 kw-hrs./bushel for Test No. 1.

## The Refrigerant System

The condensing unit required more energy per unit time in Test Nos. 3 and 4 because of the higher refrigerant flow used in these tests. However, higher temperatures of the drying air were obtained; and the energy requirements per bushel of grain dried were only slightly higher than the requirements for Test No. 1. It is difficult to compare Test Nos. 1, 3, and 4 because the relative humidity of the entering drying air was nearly the same for all of these tests.

The coefficient of performance (c.o.p.) for a vapor-compression cycle increases as the operating temperature of the evaporator approaches the condensing temperature. In Test No. 2 the evaporator was operated at a higher temperature and nearer the condensing

temperature than in Test No. 1. Theoretically, the c.o.p. for Test No. 2 should have been greater than the c.o.p. for Test No. 1. The average c.o.p.'s listed in Table 1 show that it was slightly lower. A higher evaporator operating temperature was obtained with the experimental equipment by restricting the suction line. The pressure drop across the restriction lowered the suction pressure in Test No. 2 below the suction pressure in Test No. 1, and the work of compression increased relative to the heating effect. Consequently, the increased c.o.p. obtainable by operating the evaporator at a temperature near the condensing temperature is not indicated by the data.

## The Grain System

The drying rate curves show that rather rapid drying was accomplished for about 50 hours of operation and that the drying rates became extremely slow as the equilibrium moisture contents were approached. After 50 hours of operation the grain was dried to within 3 1/2 to 4 1/2 percent of the equilibrium moisture percent and after 70 hours to within 2 1/2 to 3 1/2 percent of the equilibrium percent. A moisture gradient of 2 percent existed between the top and bottom layers after 50 hours of operation and after 70 hours the moisture gradient was reduced to approximately 1 percent. Although this slowing up of the drying process was due in part to the characteristics of drying at low moisture contents, the major cause can be attributed to the fact that as grain and air approach equilibrium the rate of drying decreases rapidly.

It is evident that to attempt to dry grain completely to the equilibrium moisture content corresponding to the entering air condition requires the expenditure of an excessive amount of energy.

The drying rate curves for Test No. 3 indicate that drying started in all layers of grain after only a few hours of operation. The grain used in this test was the same grain that had been dried in Test No. 2. It was reconditioned for Test No. 3 by forcing high humidity air through the grain. Steam was introduced into the air stream; and by controlling the amount of steam, the relative humidity of the air could be maintained at the desired value. After the steam was shut off, air was circulated through the grain for a few hours to cool the grain and to reduce the moisture gradient. Possibly the drying characteristics of this reconditioned grain were not the same as the drying characteristics of the naturally wet grain used in the other tests.

The tests show that an average of 1.57 kw-hrs./bushel was required to reduce the moisture content from 17.5 to 11.5 percent (DB). On the basis of energy per pound moisture removed, the average was .553 kw-hrs./pound moisture removed. If the cost of electrical energy is known, a comparison can be made with the drying costs reported by Foster (5). Foster's data covered a variety of heated air drying tests. Using fuel oil at a cost of 14 cents per gallon and gasoline at a cost of 21 cents per gallon, Foster's drying costs ranged from .4 to 2.2 cents per pound moisture removed.

## SUMMARY

The practicality and economics of using refrigeration equipment in a closed system to dehumidify and heat air for purposes of conditioning grain are not fully established. However, the following observations and conclusions are drawn from the analysis of the research presented in this thesis.

1. Grain in a batch type drier can be dried to a uniform moisture content by conditioning the drying air with refrigeration equipment to the equilibrium conditions corresponding to the desired moisture content.

2. To reduce the time and cost of drying, the drying air should be supplied at the equilibrium conditions corresponding to a moisture content 2 to 3 percent below the final moisture content desired. It is the author's belief that in most grain drying problems the resultant moisture gradient of 1 to 2 percent will not be objectionable.

3. When using refrigeration equipment to condition the drying air to a given temperature and relative humidity, the evaporator and condenser must be thermodynamically balanced to give the desired air conditions. Furthermore, for maximum efficiency of the refrigeration system, this balance point must allow the evaporator to operate at a temperature as near as possible to the condensing temperature.

4. The energy required for operating the heat pump in a closed system to dehumidify and heat air for grain drying when

using low air flows (six cfm/bushel) is about .25 kilowatt hours per bushel per percent moisture removed.

5. In a drying system in which the air is cooled for dehumidification before it is heated, better utilization of the drying capacity of the air can be obtained in a continuous type drier. This may be an important consideration in adapting the heat pump to grain drying because of the relatively high initial cost of the heat pump.

## ACKNOWLEDGMENT

The author wishes to express gratitude to Chester P. Davis, Jr., of the United States Department of Agriculture, for his invaluable assistance and advice in directing this study.

Appreciation is expressed to Professor F. C. Fenton, Head, and to Assistant Professor Ralph I. Lipper, both of the Department of Agricultural Engineering, for their helpful suggestions.

Acknowledgment is made to the United States Department of Agriculture, to the Kansas Committee on the Relation of Electricity to Agriculture, and to the Department of Agricultural Engineering for providing the equipment and funds necessary for this study.

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APPENDIX

Calculation of air flow energy requirements.

$$\text{Air horsepower} = \frac{\text{cfm X static pressure in inches of water}}{6356}$$

$$\text{Total power input (kw)} = \frac{\text{air horsepower X .746}}{\text{fan eff. X motor eff.}}$$

Assume:

Fan efficiency = .70

Motor efficiency = .75

| | Test No. 1 | Test No. 2 | Test No. 3 | Test No. 4 |
|---|---|---|---|---|
| Air flow (cfm) | 147.2 | 157.4 | 148.4 | 158.8 |
| Static pressure (inches of water) | .13 | .12 | .12 | .12 |
| Total power input (kw) | .0043 | .0042 | .0040 | .0043 |
| Hours of operation | 83 | 93 | 74 | 73 |
| Total energy requirement (kw-hrs.) | .357 | .391 | .296 | .314 |
| Reported in Table 1 (kw-hrs.) | .4 | .4 | .3 | .3 |

Calculation of coefficient of performance.

| | Elapsed time (hrs.) | $t_1$* | $p_1$ | $h_1$ | $t_2$ | $p_2$ | $h_2$ | $t_3$ | $p_3$ | $h_3$ | $t_4$ | $p_4$ | $h_4$ | Coefficient of performance: Heating $\frac{h_2 - h_3}{h_2 - h_1}$ | Coefficient of performance: Cooling $\frac{h_1 - h_4}{h_2 - h_1}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Test No. 1 | 7 | 64.6 | 39.7 | 86.7 | 187.2 | 169.7 | 102.0 | 92.6 | 166.7 | 29.0 | 25.3 | 40.7 | 29.0 | 4.77 | 3.77 |
| | 20 | 74.0 | 41.2 | 88.0 | 204.0 | 190.7 | 104.5 | 102.0 | 188.7 | 32.0 | 27.0 | 42.3 | 32.0 | 4.39 | 3.39 |
| | 68 | 78.0 | 39.7 | 88.7 | 198.5 | 187.7 | 103.5 | 99.4 | 181.7 | 31.5 | 25.5 | 41.7 | 31.5 | 4.87 | 3.87 |
| Test No. 2 | 7.25 | 21.4* | 36.7 | 79.5 | 129.1 | 154.7 | 93.0 | 88.7 | 152.7 | 28.5 | 51.1 | 65.7 | 28.5 | 4.78 | 3.78 |
| | 20 | 21.4* | 36.7 | 79.8 | 148.1 | 183.7 | 95.0 | 91.4 | 158.7 | 29.0 | 50.8 | 64.7 | 29.0 | 4.34 | 3.34 |
| | 71 | 21.4* | 36.7 | 79.4 | 136.0 | 166.7 | 93.5 | 91.6 | 155.7 | 29.0 | 51.3 | 66.2 | 29.0 | 4.57 | 3.57 |
| Test No. 3 | 7.5 | 33.9 | 46.7 | 82.0 | 166.6 | 184.7 | 98.0 | 100.3 | 177.7 | 32.0 | 34.8 | 48.9 | 32.0 | 4.12 | 3.12 |
| | 19.5 | 48.8 | 47.7 | 84.2 | 186.2 | 200.7 | 101.0 | 105.6 | 194.7 | 33.0 | 35.8 | 49.7 | 33.0 | 4.05 | 3.05 |
| | 74 | 55.7 | 50.7 | 85.0 | 189.0 | 211.7 | 101.3 | 110.2 | 204.7 | 34.5 | 35.9 | 52.7 | 34.5 | 4.10 | 3.10 |
| Test No. 4 | 7 | 35.5* | 47.7 | 80.4 | 140.0 | 182.7 | 93.5 | 104.6 | 180.7 | 32.4 | 52.5 | 65.7 | 32.4 | 4.66 | 3.66 |
| | 21 | 60.0 | 49.7 | 85.7 | 185.9 | 214.7 | 100.5 | 114.0 | 212.7 | 34.6 | 50.5 | 64.7 | 34.6 | 4.45 | 3.45 |
| | 73 | 33.0 | 42.7 | 82.2 | 150.0 | 169.7 | 96.0 | 96.5 | 166.7 | 30.5 | 56.4 | 60.2 | 30.5 | 4.74 | 3.74 |

*This temperature for the vapor compression cycles where the refrigerant entered the compressor with an unknown quality was obtained by assuming an entropy increase of .0050 during compression. This was an average increase of entropy for the compression cycles of other tests where the refrigerant entered the compressor in a superheated condition.

Symbols:

Letters
- t = temperature (°F)
- p = pressure (lbs./sq. in. abs.)
- h = enthalpy (Btu./lb. of refrigerant)

Subscripts
- 1 = condition before compression
- 2 = condition after compression
- 3 = condition after condensation
- 4 = condition after expansion through expansion valve

# A LABORATORY INVESTIGATION OF THE ADAPTABILITY OF THE HEAT PUMP TO BATCH DRYING OF SHELLED CORN

by

GENE CLERE SHOVE

B. S., Kansas State College of Agriculture
and Applied Science, 1952

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AN ABSTRACT OF A THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Agricultural Engineering

KANSAS STATE COLLEGE
OF AGRICULTURE AND APPLIED SCIENCE

1953

In recent years there has been an increased interest in the agricultural potentialities of the heat pump. Several writers have suggested the possibility of using the heat pump to condition farm crops for storage. Early harvesting of high moisture content grain has resulted in a need for an economical method of drying the grain before it can be placed on the market or stored.

High moisture content grain is usually dried by forcing heated air through the grain. Most of the heated air driers now available are of the combustion type using oil or a similar fuel for the heat source. These driers operate satisfactorily but involve certain thermal inefficiencies. The principal loss is the exhausting of latent and sensible heat to the ambient air by way of the drying air which is forced through the grain. The heat pump operating in a closed system where the drying air is recirculated can reclaim for further use this latent and sensible heat which is normally lost. Also the heat pump operating in a closed system will operate independently of ambient temperature and relative humidity. The temperature and relative humidity of the drying air within the system can be controlled; and by supplying the drying air at the equilibrium condition corresponding to the desired final moisture content, the entire grain mass can be dried to a uniform moisture content. The heat pump also offers the advantage of automatic operation, and the fire hazard which exists with the combustion type driers is eliminated.

The investigation was initiated to study the possibilities of drying grain with the heat pump. An experimental laboratory installation was built, and drying tests were made on shelled corn. This installation was a closed system in which the drying air was recirculated. The moist air from the grain was passed over the cold evaporator where the air temperature was reduced to its dew point and moisture removed. After leaving the evaporator, the air passed over the condensing unit where it was heated and then passed on to the grain bed for completion of the cycle. A fan placed in the system maintained an air movement of about six cfm/bushel.

Instruments were installed to record and indicate the various temperatures, pressures, weights, and energy requirements. A grain probe was used to obtain grain samples for moisture content determinations. Oven dried samples were used for the moisture content calculations, and for immediate checking during the tests a portion of each sample was put through a Tag-Heppenstall moisture meter.

A preliminary test was made to determine the limitations of the equipment. Four tests were then conducted and analyzed, and the following conclusions were drawn.

Refrigeration equipment can be successfully used to dry grain to a uniform moisture content by conditioning the drying air to the equilibrium condition corresponding to the desired moisture content. However, as the grain and air approach equilibrium, the rate of drying decreases rapidly. If an attempt is made to

dry grain completely to the equilibrium moisture content corresponding to the entering air condition, an excessive amount of energy is required. To reduce the time and cost of drying, it is desirable to supply the drying air at the equilibrium condition corresponding to a moisture content 2 to 3 percent below the final moisture content desired. The small moisture gradient that results in all probability will not be objectionable.

When using refrigeration equipment to dehumidify and heat air for grain drying, the evaporator and condenser must be thermodynamically balanced to give the desired air condition. Also, for maximum efficiency of the refrigeration system, the evaporator must operate at a temperature as near as possible to the condensing temperature.

An average of 1.57 kw-hrs./bushel was required to reduce the moisture content from 17.5 to 11.5 percent (DB); or on the basis of energy per pound moisture removed, the average was .553 kw-hrs./pound moisture removed. It appears that the cost of operating the heat pump to condition air for grain drying will compete favorably with other methods of drying. However, the relatively high initial cost of the equipment may limit its applications.